

## STEAM TURBINES

*These efficient machines are the principal means of converting the heat energy released by fossil and nuclear fuels into the kinetic energy needed to drive power generators and large ships*

by Walter Hossli

The steam turbine ranks with the internal-combustion engine as one of the major achievements of mechanical engineering in the 19th century. Steadily increased in size, reliability and efficiency, steam turbines now account for more than 75 percent of the electric power generated in the world (most of the rest is hydroelectric) and propel most of the biggest and fastest ships. Between 15 and 20 percent of the fossil fuel consumed in the U.S. and Western Europe - and essentially all the fuel now consumed in nuclear power plants - has only one purpose: to evaporate and superheat water that then passes through a steam turbine. For all their size and ubiquity steam turbines are among the least understood products of the mechanical age; they are seldom located where they can be seen by the public and the complexities of their design are familiar only to specialists. It took the recent failure of the turbines in the new Cunard liner Queen Elizabeth 2 to remind people not only that these great machines exist but also that their design and construction is an exercise in high technology.

In principle a steam turbine is simplicity itself. It is a pinwheel driven by high-pressure steam rather than by air. It basically consists of a rotor from which project several rows of closely spaced buckets, or blades. Between each row of moving blades there is a row of fixed blades that project inward from a circumferential housing. The fixed blades are carefully shaped to direct the flow of steam against the moving blades at an angle and a velocity that will maximize the conversion of the steam's heat energy into the kinetic energy of rotary motion. Because the steam's temperature, pressure and volume change continuously as it progresses through the turbine, each row of blades has a slightly different length, and in certain parts of the turbine the twist of the blade is usually varied along the length of the blade, from root to tip. At the inlet of the turbine the blades are stubby, with little or no twist; at the outlet the blades are much longer and the twist is pronounced.

In a typical modern power plant steam leaves the boiler, after being super-heated, at a pressure of 2,500 pounds per square inch and a temperature of 1,000 degrees Fahrenheit [see Fig. 5]. As it enters the high- pressure turbine (the first unit in a cascade of three or four, usually working on a single shaft) a pound of steam occupies about .3 cubic foot. A short time later, when it leaves the low-pressure turbine (the last unit in the cascade), a pound of steam occupies more than 300 cubic feet, a thousand fold expansion.

The steam enters the high-pressure turbine at a velocity of about 150 feet per second and is immediately accelerated by expansion in the first row of fixed blades. In the

low-pressure turbine, steam velocity and blade velocities can exceed 1.5 times the velocity of sound under the existing steam conditions. When the steam finally leaves the low-pressure turbine, its velocity is about 600 to 1,000 feet per second. Thus the turbine engineer must design "airfoils" (more accurately blade profiles) able to operate efficiently over a range of velocities roughly equivalent to the range encountered by the designers of a supersonic airplane.

It is now 85 years since the British engineer Charles A. Parsons conceived the modern steam turbine and energetically began pushing its practical application. In an age commonly regarded as ultraconservative the steam turbine was developed and put to large-scale use with remarkable speed - a speed not equaled later by its descendant, the aircraft gas turbine, in an age thought to be much more receptive to innovation. Parsons demonstrated his first steam turbine in 1884. Seven years later he saw the value of adding a condenser to exploit the turbine's distinctive capacity for utilizing the energy of low-pressure steam down to a near-vacuum. That same year, 1891, the first steam turbines were harnessed to generate electricity.

In 1897 Parsons installed a 2,100 - horsepower turbine in the *Turbinia*, a vessel of 44 tons, and astonished the British Admiralty by driving it at 34 knots through a parade of warships in a naval review at Spithead. The fastest ship yet built, the *Turbinia* easily out raced the entire fleet, which was equipped with conventional reciprocating steam engines. By 1904 the Admiralty had installed one of Parsons' turbines in a cruiser, the *Amethyst*, and a year later decided to abandon reciprocating engines and put Parsons turbines of 23,000 horsepower in the new Dreadnought class of battleships and turbines of 41,000 horsepower in the 28-knot *Invincible* class of cruisers. In 1907 the Cunard line launched the *Lusitania* and the *Mauretania*, each equipped with Parsons turbines developing 70,000 horsepower. Nearly twice as powerful as the largest reciprocating engines ever employed for ship propulsion, the turbines drove the two Cunard liners at a record speed of 25 knots. Only 16 years had elapsed since Parsons had thought to add a condenser to a turbine of a few hundred horsepower.

His accomplishments made such an impact on the British that they were recounted in much detail in the famous 11th edition of the *Encyclopedia Britannica*, published in 1910-1911. (The same edition does not mention Einstein's special theory of relativity, published in 1905 or Planck's concept of the quantum of energy, published five years earlier.) The author of the entry on steam turbines has no hesitancy in describing "the invention of the steam turbine [as] the most important step in steam engineering since the time of Watt."

Steam turbines for ship propulsion have not grown very much since the days of the *Lusitania*. The turbines of the *Queen Elizabeth 2* are designed to produce 110,000 horsepower. Turbines of only 40,000 horsepower enable the largest tankers, vessels of more than 300,000 tons, to travel at 13 knots. The biggest increase in turbine size has been in the field of electric-power generation, and the end of this evolutionary trend is not in sight. In 1900 the first steam turbine used for power generation in continental Europe had a rating of 250 kilowatts; it was built by Brown, Boveri & Company Limited of Switzerland, the firm with which I am affiliated. (For purposes of comparing electric-power turbines and marine turbines, one kilowatt is about 1.3 horsepower.) Brown, Boveri is now building for the Tennessee Valley Authority the world's two largest steam turbine sets; each will have an output of 1,300,000 kilowatts,

or 1,300 megawatts [see Figs. 11 & 12]. Our projections show that well before the end of the century single units of 2,000 megawatts will be required for the most efficient generation of power.

For propulsion of big ships and power generation, where large unit output and maximum efficiency are essential, the steam turbine is unchallenged. Diesel engines have a comparable efficiency but the largest units so far built are limited to about 30,000 kilowatts. Gas turbines go up to 100,000 kilowatts, but their efficiency is somewhat lower. At the other end of the size range, steam is again being considered as a power medium for automobiles; it remains to be seen, however, whether turbines or piston units will be more successful. The incentive is a reduction in air pollution. Nearly complete combustion can be achieved when fuel is used to fire a boiler instead of being burned in the cylinder of an engine.

Present-day applications of land-based steam turbines are by no means limited to electric power production. In many industrial plants, such as refineries, sugar mills, paper mills and chemical plants, low-pressure steam is needed in large quantities for process purposes. Instead of producing this steam directly in a low pressure boiler it is advantageous to install a high-pressure boiler at little extra cost and to drop the steam to the desired level by expanding it through a "back pressure" turbine whose outlet pressure corresponds to the pressure needed for process steam. (If steam at two pressures is desired, a second turbine can be installed ahead of the back-pressure turbine so that steam can be withdrawn between the two units as well as after the second.) When this is done, electric power can be produced as a by-product at almost no cost. In most cases, of course, an external power supply is needed to handle fluctuating electric loads. It is also common in industry to use steam turbines as a direct power source for rotating machinery, such as compressors, blowers and pumps. All told, the worldwide demand for steam turbines requires the production of units whose annual combined output exceeds 50,000 megawatts, or 65 million horsepower.

The development of practical steam engines occupied many minds in the 18th and 19th centuries. The first successful steam engine had been introduced in 1698 by Thomas Savery. It was a crude affair in which steam was condensed alternately in two chambers, creating a vacuum that could be used to draw water from mines. In 1705 Thomas Newcomen built the first practical steam engine with a piston. In 1763 James Watt began making his contributions, and in 1781 he was the first to patent methods for converting the reciprocating motion of a piston steam engine to rotary motion. With this advance the steam engine finally became a versatile prime mover. By 1804 Richard Trevithick had built the first steam-driven locomotive.

During the remainder of the 19th century piston steam engines were steadily improved by many inventors; however, they saw the advantages that would result if steam could be used directly to produce rotary motion by means of some kind of turbine. Many devices were built in crude imitation of waterwheels. It remained for Parsons to recognize that what was needed was a device with many rows of buckets in which a small amount of the steam's kinetic energy would be extracted with high efficiency at each of many successive stages. Whereas the piston steam engine exploited only the pressure and temperature of steam as it came from a boiler, the steam turbine used some of the pressure to create high-velocity jets, whose energy was then absorbed by the rotating blades.

Early in the turbine's history two concepts of blade arrangement were developed, each with its champions. Parsons favored what became known as reaction blading. Some of his competitors adopted impulse blading [see Figs. 6 & 7]. In the reaction turbine the fixed blades and the moving blades that constitute one stage are practically identical in design and function; each accounts for about half of the pressure drop that is converted to kinetic energy in the entire stage. In the fixed blades the pressure is harnessed to increase the velocity of the steam so that it slightly exceeds the velocity of the moving blades in the direction of rotation. In the moving blades the pressure drop is again used to accelerate the steam but at the same time to turn it around (with respect to the blades), so that its absolute tangential velocity is almost zero as it enters the next bank of stationary blades. Thus thrust is imparted to the moving blades as the steam's absolute tangential velocity is reduced from slightly above blade speed to approximately zero. An imaginary observer moving with the steam could not tell whether he was passing through the fixed blades or the moving ones. As he approached either type of blade it would appear to be nearly motionless, but as he traveled in the channel between blades his velocity would increase steadily until he reached their trailing edges, which would then seem to be receding rapidly.

In the impulse turbine the fixed blades are quite different in shape from the moving ones because their job is to accelerate the steam until its velocity in the direction of rotation is about twice that of the moving blades. The moving blades are designed to absorb this impulse and to transfer it to the rotor in the form of kinetic energy. In this arrangement most of the pressure drop in each complete stage takes place in the fixed blades; the pressure drop through the moving blades is only sufficient to maintain the forward flow of steam. The amount of energy transferred to the rotor in each stage is proportional to the change in absolute steam velocity in the direction of rotation. This is the value labeled  $\Delta C_u$  in the illustrations Figs. 6 & 7. It turns out that the value is about twice as high for impulse blading as it is for reaction blading. This means, in turn, that an impulse turbine will need fewer stages for the same power output than a reaction turbine; the efficiency, however, will be about the same for both types.

This being the case, one would expect impulse blading to have carried the day. Not so. As often happens in engineering, a design that seems clearly superior can present secondary problems of such magnitude that the choice between the alternatives becomes very nearly equal. In turbine design one of the major secondary problems is providing seals to keep the steam from leaking through the narrow spaces between the rotor and the stator. In impulse blading the complete expansion in each stage takes place in the fixed blades. It is thus desirable to place the seals on as small a diameter as possible. This has led to a turbine design known as the diaphragm type [see Fig 8]. Because the pressure differential is large the diaphragm needs considerable space in the axial direction. Therefore the width of the fixed blade must be made larger than it would otherwise have to be. A circumferential shroud is often placed around each ring of moving blades.

In reaction blading the pressure drop per stage is less than it is in impulse blading; moreover, it is divided equally between fixed and moving blades. Thus both blades can be fitted with similar seals, and the seals need not be as effective as those needed on the fixed blades in impulse blading. The result is a drum turbine [see Fig 9]. Another advantage of the reaction turbine is that the stationary and moving blades in each stage can have the same shape, which simplifies design and yields

manufacturing economies.

For more than 50 years these two kinds of turbine, the diaphragm turbine and the drum turbine, have been in competition without either type's demonstrating a distinctive advantage. Along the way the advocates of the two designs have moved somewhat away from pure reaction or pure impulse blading to adopt various compromise arrangements.

The efficiency that can be attained at each stage in a large modern turbine is quite remarkable: more than 90 percent. For large units with reheat systems an overall turbine efficiency of about 88 percent can be achieved. This is not the net thermal efficiency of the steam turbine as a heat engine, however. For this calculation one must introduce the limitations on the theoretical efficiency of heat engines first described in 1824 by Nicolas Leonard Sadi Carnot, who recognized that heat does work only as it passes from a higher temperature to a lower one. Specifically the efficiency is  $1 \text{ minus the fraction "final temperature"/"initial temperature,"}$  both values being expressed on the Kelvin, or absolute, scale. If the working fluid is initially at 1,000 degrees F. (811 degrees K,) and is finally at 100 degrees F. (311 degrees K.), the maximum theoretical efficiency is about 60 per cent  $[(1 - 311/811) \times 100]$ . Because the inherent properties of a steam cycle do not allow the heat input to take place at a constant upper temperature, the theoretical maximum efficiency is not 60 percent but closer to 53 percent. Thus the actual thermal efficiencies achieved with large reheat-steam turbines is .88 (the turbine efficiency) times 53, or some what better than 46 percent.

Today the chief point of competition among rival turbine manufacturers is not efficiency, since all guarantee comparable performance, but the capital cost of the unit. As units have become steadily larger the cost per megawatt of capacity has dropped substantially because of economies inherent in size. For example, a turbine of 125 megawatts, which was considered large 15 years ago, weighed about 2.64 tons per megawatt. The 1,300-megawatt turbine Brown, Boveri is building for the TVA will weigh less than half that amount per megawatt. The cost per megawatt is roughly proportional to weight.

In the first 50 years of the steam turbine's history (roughly 1890 to 1940) turbine design was guided mainly by the intuition and ingenuity of engineers who were never far from the shop floor. Today, with the sharp increase in power density and the use of steam at higher pressures and temperatures, turbine progress depends increasingly on scientific understanding and the skillful application of new problem-solving tools such as the electronic computer. The design of a modern turbine requires the solution of difficult problems in aerodynamics, applied mathematics, metallurgy, vibration and the physical behavior of steam, together with attention to many manufacturing problems (such as the production of complex blade contours at reasonable cost, the setting of permissible tolerances and size limitations imposed by transportation).

Rather than discuss such problems as they apply to the turbine as a whole I shall speak only of the design of the moving blades used in the last stage at the low-pressure end of the turbine. These are the longest blades in the turbine; hence the blades with the highest tip speed and the most acute vibrational problems. They are also the blades most subject to erosion from water droplets, which tend to appear as

the steam approaches the turbine exit just before entering the condenser.

In designing such blades aerodynamic calculations reach a complexity exceeding that met in other sections of the turbine, where pressures are higher and flow patterns somewhat simpler. In passing through the low-pressure turbine the volume of steam expands about 100 times, with the result that the path of the steam has a strong radial component in addition to its forward component. The calculations must therefore deal with steam flow in three dimensions. An equilibrium condition must be maintained between the centrifugal forces that throw the steam outward and the distribution of radial pressure. In a very poor design the moving blades will tend to act as a compressor. The solution to this complicated flow problem involves differential equations that can be solved only with a large-capacity computer.

The results of these initial calculations provide the designer with a preliminary concept of the blade cascade, indicating the number of stages and the inlet and outlet angles of the blades in each stage. The next step is to design blades with high efficiency from root to tip. Toward this end the designer chooses profiles whose characteristics are known either from calculation or actual measurement. Even near the root, where the steam flow is subsonic, existing blade profiles may be unsatisfactory; new profiles must then be developed and evaluated. In the outer third of the blade, where the flow passes from subsonic to supersonic, things become more difficult. In calculating the line where sonic velocity is reached, mathematical equations tend to become unstable, meaning that they fail to provide a reasonable result. One must then use a combination of empirical and analytical methods. The critical passage from subsonic to supersonic velocity, known as the sonic line, must be verified by experiments in a water tank or in a rotating cascade in which the fluid is air.

In the supersonic region calculation becomes simpler again. Good methods exist for predicting the location of regions, called Mach lines, where minor disturbances will cause sudden but small changes in steam velocity and pressure. One can also compute shock lines; the lines along which high-compression waves will develop. Water-tank methods, in which water passes swiftly between model blades, provide an excellent visual representation of flow patterns and shock waves [see Fig 9]. Under suitable conditions flowing water behaves like a supersonic flow of air or steam. From such studies, with the help of scaling laws, one can readily determine the supersonic flow pattern as well as the sonic line. The results can be checked by passing air through a rotating cascade of blades and using optical methods to disclose the flow patterns. A critical comparison of the three methods - mathematical analysis, water tank flow and airflow in a cascade - enables the designer to make a sound prediction of actual steam-flow conditions. In this way the aerodynamic side of the problem can be solved.

**T**he designer is now ready to sketch a preliminary configuration for the blades in the last stage of the low-pressure turbine. Because they are highly twisted these blades are subject to unusual stresses. Under centrifugal force a last-stage blade may untwist more than seven angular degrees. To keep material stresses under control the positioning of the centers of gravity in various sections of the blade and also of the centers of inertia is very important.

Not the least of the engineer's problems is to design a root that will hold the blade in the rotor against a centrifugal force of more than 250 tons. The space available for

transmitting this force to the rotor body is very limited. Tiny strain gauges and polarized-light studies of transparent plastic models provide information about stress distribution, and sample roots are stressed to destruction. Such tests can shed light on the rupture mechanism and indicate the safety factor expected when the blade is run at normal speed.

Vibration can be particularly troublesome in long blades. The failure of blades in the turbines of the Queen Elizabeth 2 has been traced to vibrations at resonance frequencies, attributable to faulty design. In designing the last-stage blades one must calculate resonance frequencies and where the nodal points will occur. The calculation method accounts for centrifugal and torsional frequencies as well as for their coupled effect. The calculated values must then be verified by static and rotational tests; close agreement is usually obtained between calculation and experiment [see Fig 15].

Even these tests are not sufficient. The distribution of steam pressure and velocity acting on the blade may vary considerably with the load, so that additional tests must be made. This is usually done in a test turbine scaled down from a complete low-pressure turbine. One can study how the last-stage blading is influenced by stages preceding the last stage under varying load conditions and also examine the back effects produced by the exit diffuser. Reliability of the last-stage blade is so important that further tests are run on turbines in commercial operation. Tiny strain gauges are attached to the blade, and their measurements are transmitted by radio.

In addition to determining resonance frequencies up to the sixth or seventh mode of vibration, it is necessary to check carefully still higher modes in a range near the exciting frequencies produced by the stationary blades. There are evidently flow disturbances, similar to wakes, behind each stationary blade that are capable of inducing resonances characteristic of a vibrating plate. Fortunately this phenomenon is nearly independent of centrifugal effects, so that stationary tests are sufficient. Laser holography has recently been used to record the vibration pattern in such tests.

The metallurgical requirements for a last-stage blade are, as one might expect, demanding. The steel should have a high yield point combined with good ductility in order to withstand high centrifugal stresses. It should resist fatigue as well as erosion and chemical- and stress-corrosion. (In normal operations one can expect small quantities of corrosive chemicals to be entrained occasionally in the steam.) The steel should also have high damping characteristics to minimize vibration. Finally, the steel should lend itself to machining or forging; more exotic means of shaping a blade are usually very costly.

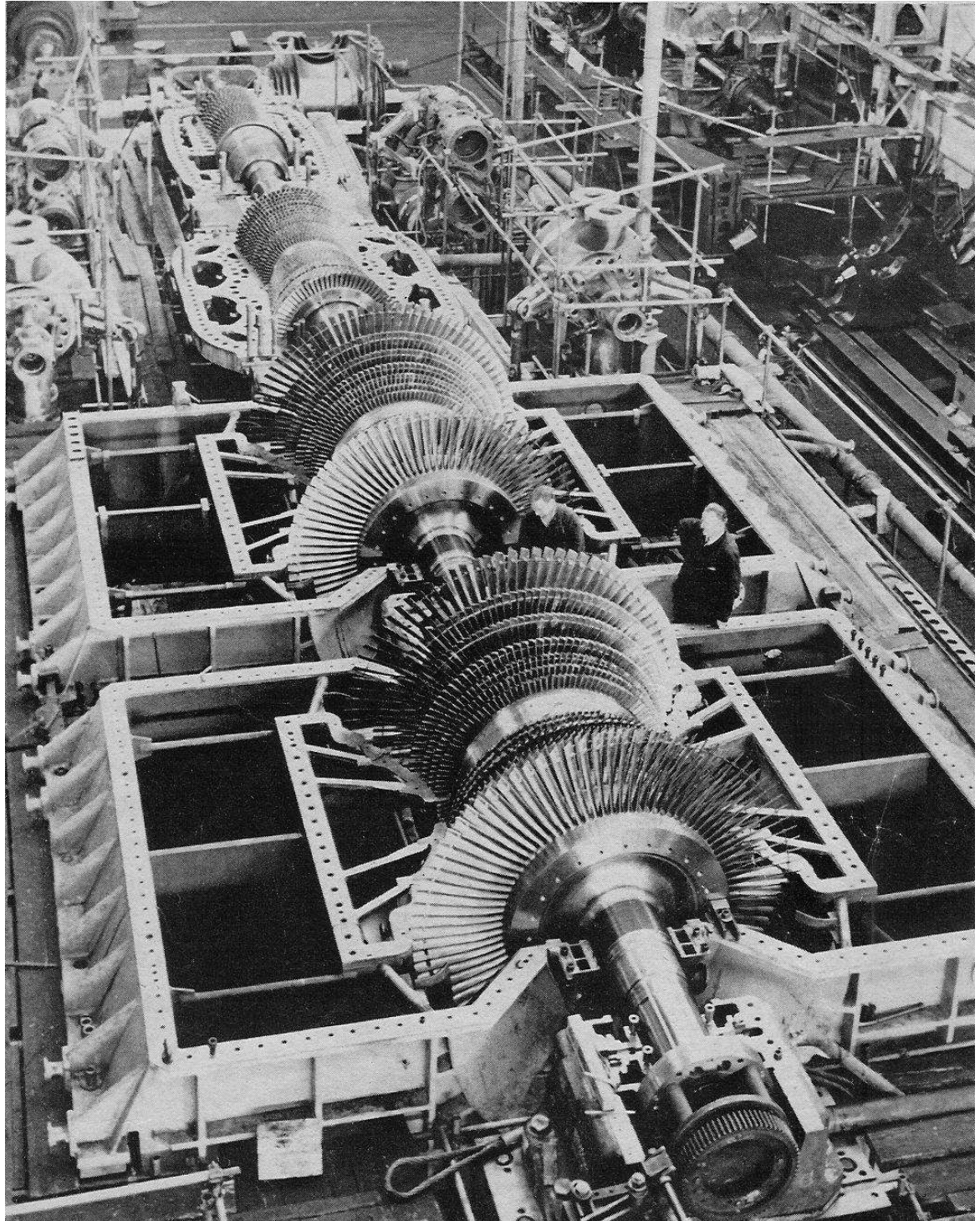
If the steel itself does not possess sufficient resistance to erosion, the blade can be hardened or surfaced with an erosion-resistant hard alloy such as Stellite. Much study has been devoted to the mechanism by which wet steam causes erosion. Because the steam passing through the low-pressure turbine is wet, drops of water tend to collect on the trailing edge of the stationary blades. The drops are then torn away by the passing steam and hurled against the moving blades producing tiny pits. The designer's task is to see that pitting does not impair the efficiency and reliability of last-stage blades over the 20-year minimum lifetime expected of a steam turbine. Blades in the high-pressure turbine, where the temperatures are highest, are virtually free from erosion.

I have described in some detail the development of last-stage blades not only because their design involves a representative range of engineering problems but also because they place one of the limits on the size to which individual turbine sets can grow. For maximum economy high-pressure, intermediate- pressure and low-pressure turbines should be arranged in a single line. The more steam per hour that enters the high-pressure end of the set, the larger the blades required at the low-pressure end to handle the much expanded volume of steam. "Half-speed" turbines (1,800 revolutions per minute instead of 3,600) have been introduced to provide large output in a single turbine set, but their weight and cost, megawatt for megawatt, are higher than those for full-speed turbines. To hold costs in line, therefore, it is imperative for manufacturers to learn to design full-speed turbines with larger low-pressure last stages than any built so far. These larger units will be needed to meet the growing demand for electric power at a steadily reduced cost per kilowatt-hour.

*This is a reproduction through picture scanning and text OCR of an April 1969 article appearing in Scientific American. Changes to formatting have been made and the pictures assigned Fig. X numbers. The text was modified to use Fig. X references when referencing pictures and diagrams.*

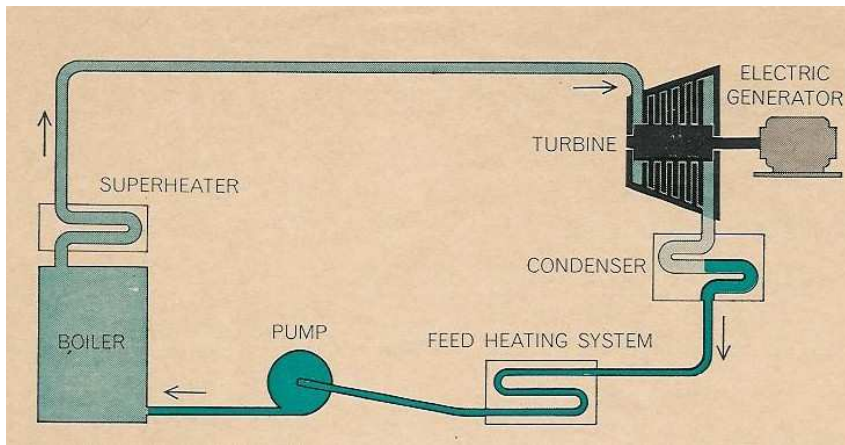
*Stephen Patterson*



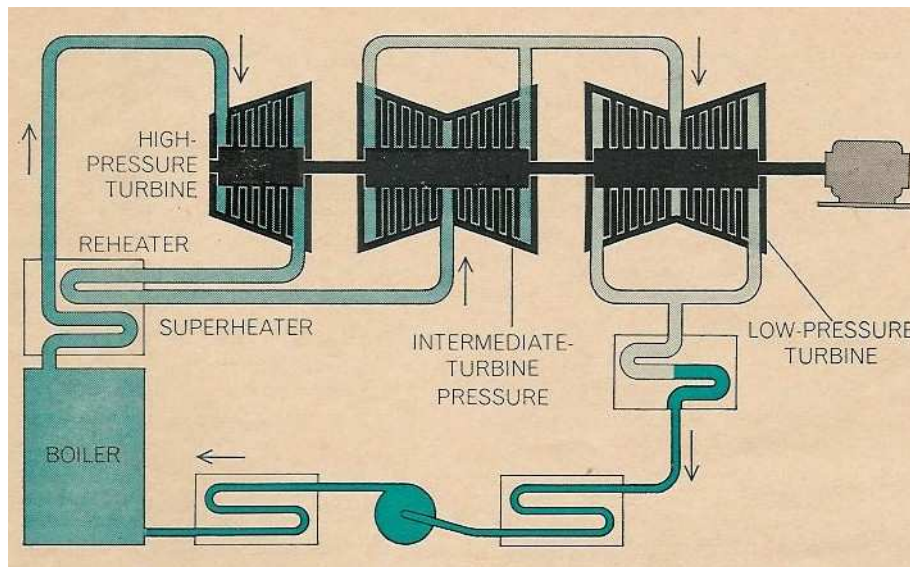


**Fig. 1**

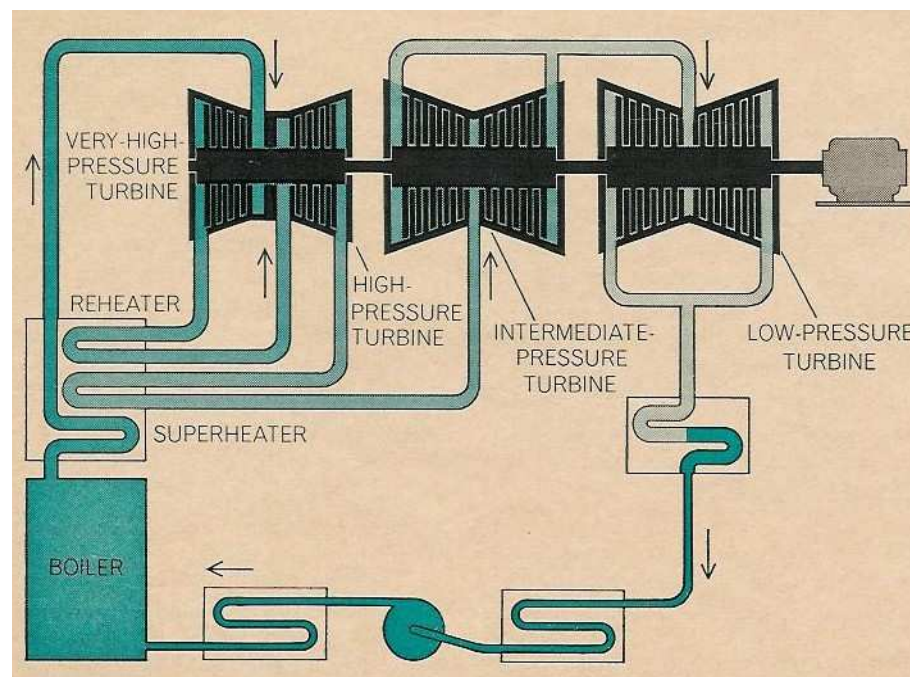
*ANATOMY OF A STEAM TURBINE is exposed in the shops of Brown, Boveri & Company Limited near Zurich. When connected to an electric generator, this turboset, as it is called, is capable of producing 320 megawatts of electricity, equivalent to approximately 400,000 horsepower. It thus ranks among the larger sets built with a single shaft. Steam enters the turboset at a pressure of 2,640 pounds per square inch and a temperature of 977 degrees Fahrenheit. After passing through the high-pressure turbine, located at the extreme rear, the steam is reheated to 977 degrees and then enters a double-flow intermediate-pressure turbine, located in the housing next in line. The steam emerges from this unit divided into two streams and passes, without further reheating, into two double-flow low-pressure turbines, which are located in the large boxlike housings in the foreground. This division of flow is needed to accommodate the thousand fold expansion in volume that takes place as the steam travels through the turbine from inlet to outlet.*



**Fig. 2**  
**SIMPLE**  
**TURBOGENERATOR** is used where power demand is light or fuel is cheap. Steam passes through a single turbine. Steam pressure is limited to about 1,500 pounds per square inch, output to 100 megawatts. Maximum thermal efficiency is around 37 percent.

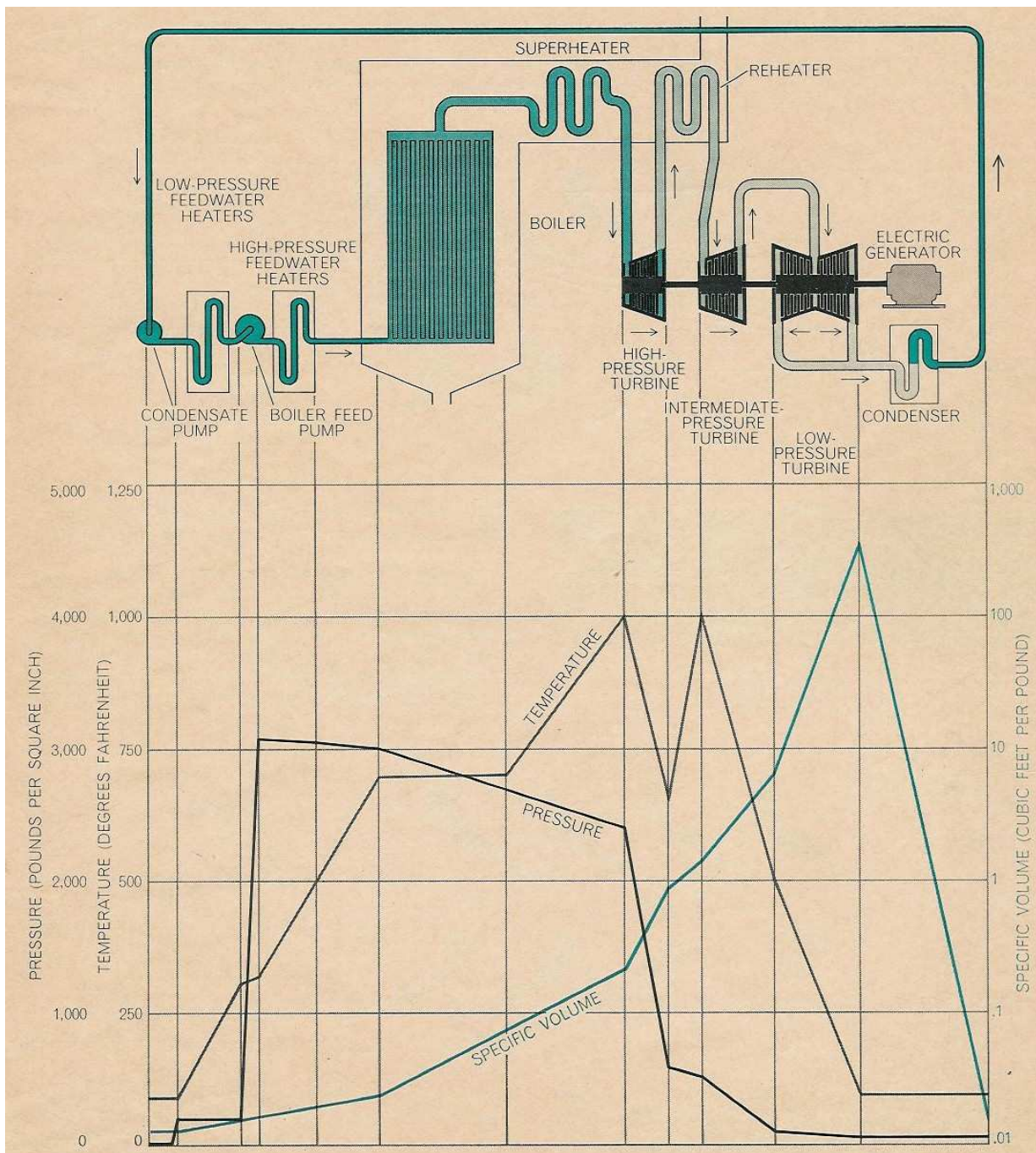


**Fig. 3**  
**SINGLE-REHEAT**  
**TURBOSET** is usually preferred when the demand for power exceeds 100 megawatts and fuel costs make a thermal efficiency above 40 percent desirable. After steam leaves the high-pressure turbine it is reheated and returned to the double-flow intermediate-pressure turbine. From there it passes to the double-flow low-pressure turbine.



**Fig. 4**  
**DOUBLE-REHEAT**  
**TURBOSET** provides the highest power output and the highest efficiency, around 47 percent. In all three systems feed water is heated by steam bled from the turbines. Cooling water for the condensers often comes from a nearby river, lake or ocean.

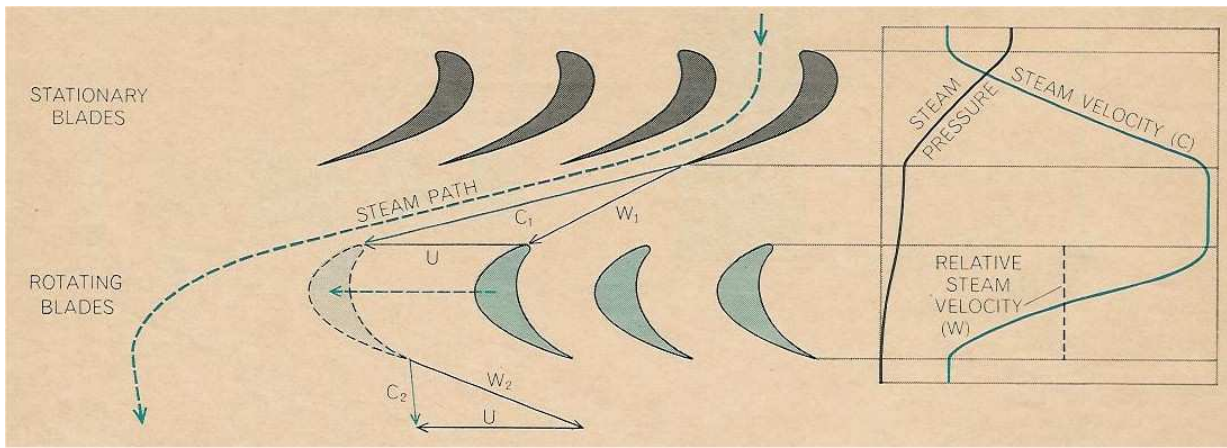




**Fig. 5**

STEAM TEMPERATURE, PRESSURE AND VOLUME are the critical factors in the design of a steam power plant. The curves show how these three factors vary throughout a typical system. Pressure is Highest at the exit of the feedwater pump leading to the boiler. At the entrance to the high-pressure turbine the pressure has dropped somewhat to around 2,400 pounds per square inch. Thereafter it falls rapidly as it passes through the turbine cascade. The steam temperature is raised to 1,000 degrees F. in the super-heater and again in the reheater, finally plunging to about 80 degrees F. as it leaves the low-pressure turbine. The specific volume of the steam (color) varies over the greatest range and is therefore plotted on a logarithmic scale. At the inlet to the high-pressure turbine one pound of steam occupies about .3 cubic foot. When the steam leaves the turbine cascade, it occupies about 300 cubic feet.

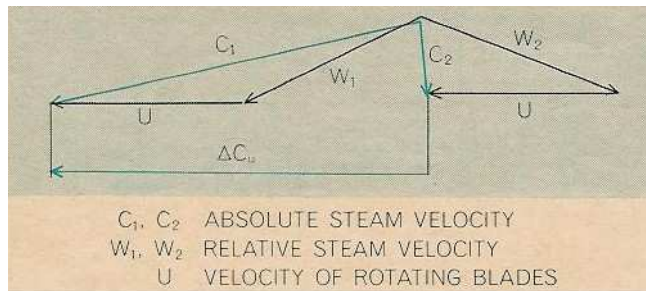




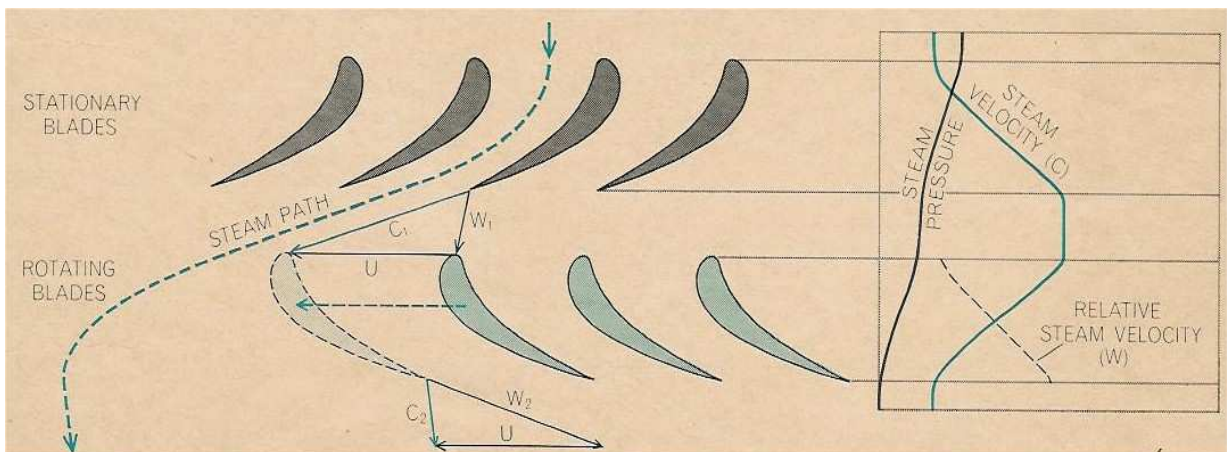
**Fig. 6**

**IMPULSE BLADING** is one of two general methods for extracting kinetic energy from steam in a turbine. In the stationary blades steam is accelerated to a velocity ( $C_1$ ) about twice that of the moving blades ( $U$ ). Velocity is obtained at the expense of pressure (curves at right). The moving blades extract kinetic energy from the fast-moving steam, so that it leaves with essentially no tangential component of velocity ( $C_2$ ). In passing through one row of fixed blades and one row

of moving blades, called a stage, the amount of energy transferred to the rotor is proportional to the change in absolute steam velocity,  $\Delta C_u$  (vector diagram).



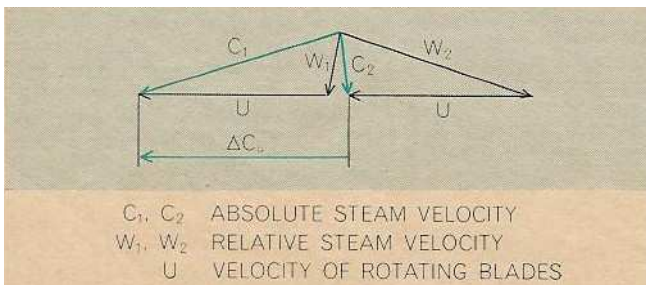
$C_1, C_2$  ABSOLUTE STEAM VELOCITY  
 $W_1, W_2$  RELATIVE STEAM VELOCITY  
 $U$  VELOCITY OF ROTATING BLADES



**Fig. 7**

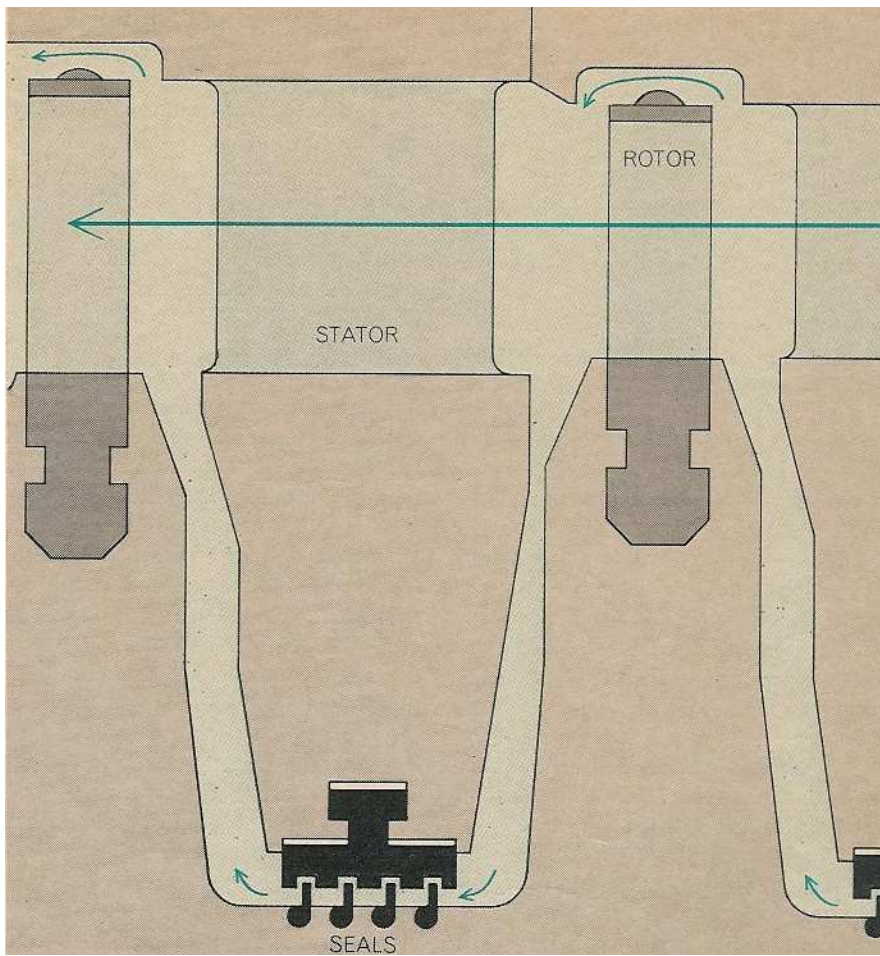
**REACTION BLADING** is the other concept used in steam turbine design. Here the pressure drop per stage is equally divided between fixed and moving blades (curves at right). In the fixed blades steam is accelerated to a velocity ( $C_1$ ) only slightly greater than that of the moving blades ( $U$ ). Continued expansion of the steam in the moving blades provides thrust and gives the steam a relative velocity ( $W_2$ ) equal and opposite to its former absolute

velocity ( $C_1$ ). In reaction blading the energy,  $\Delta C_u$  transferred to the rotor in a single stage (vector diagram) is only about half that transferred by impulse blading. Efficiencies, however, are comparable.

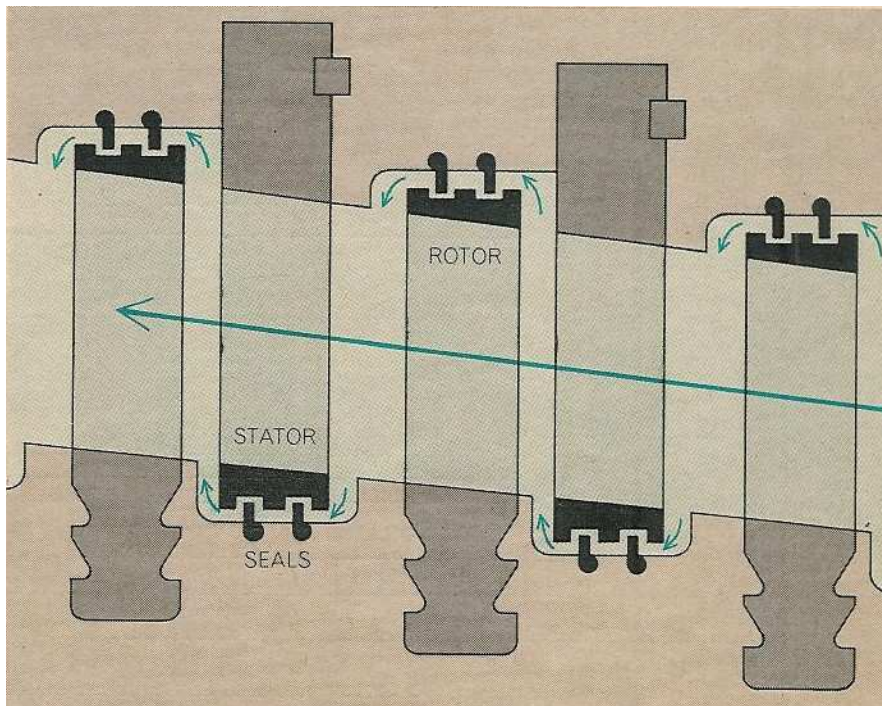


$C_1, C_2$  ABSOLUTE STEAM VELOCITY  
 $W_1, W_2$  RELATIVE STEAM VELOCITY  
 $U$  VELOCITY OF ROTATING BLADES

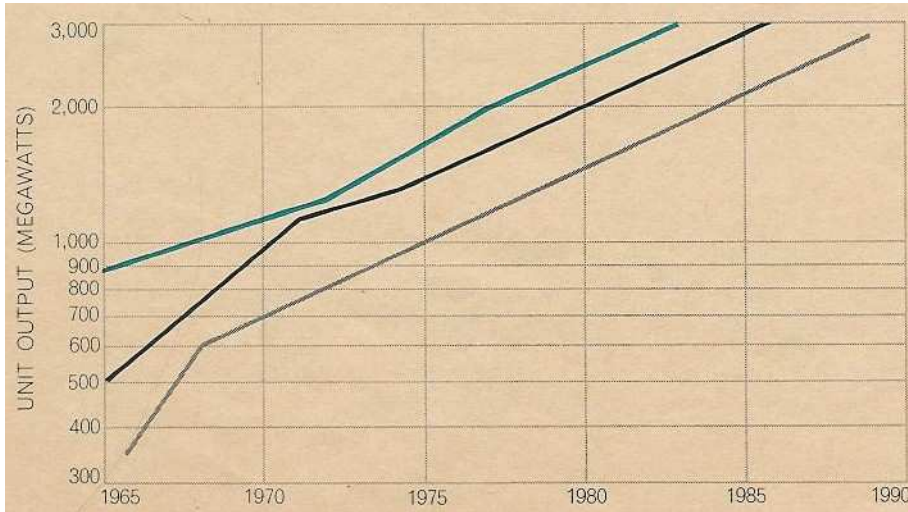




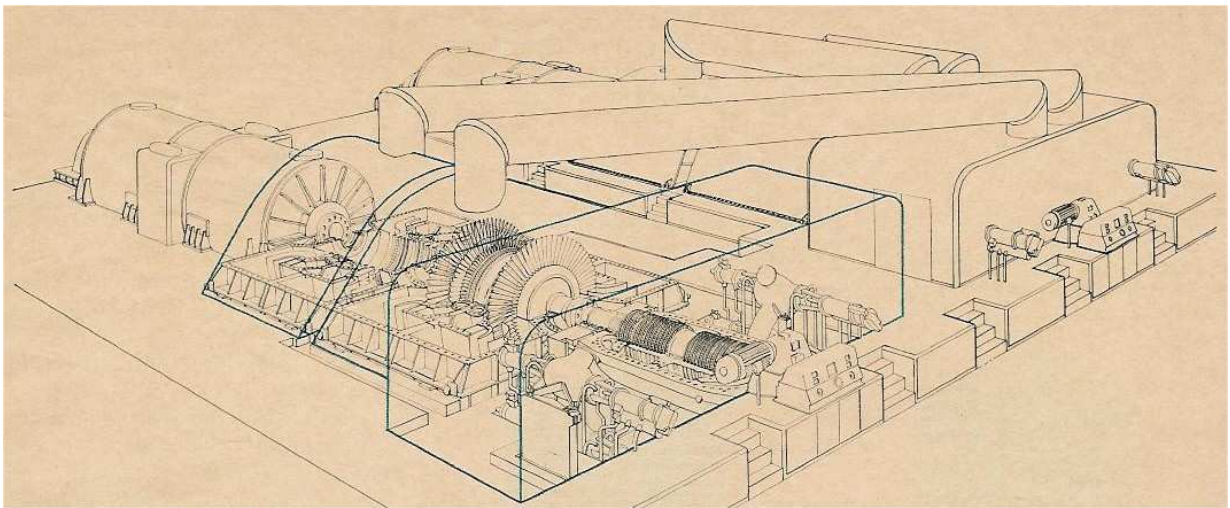
**Fig. 8**  
**DIAPHRAGM TURBINE** provides effective seals (black) in turbines with impulse blading, where the complete expansion of steam per stage occurs in the stator, or fixed, blades. This design allows seals to be placed on as small a diameter as possible. The moving blades are usually covered by a circumferential shroud, which may also carry seals (not shown).



**Fig. 9**  
**DRUM TURBINE** offers the simplest arrangement of seals (black) when reaction blading is used. Seals on rotor and stator blades can be identical because the pressure drop across them is equal. Also the seals can be simpler than those needed for impulse blading.



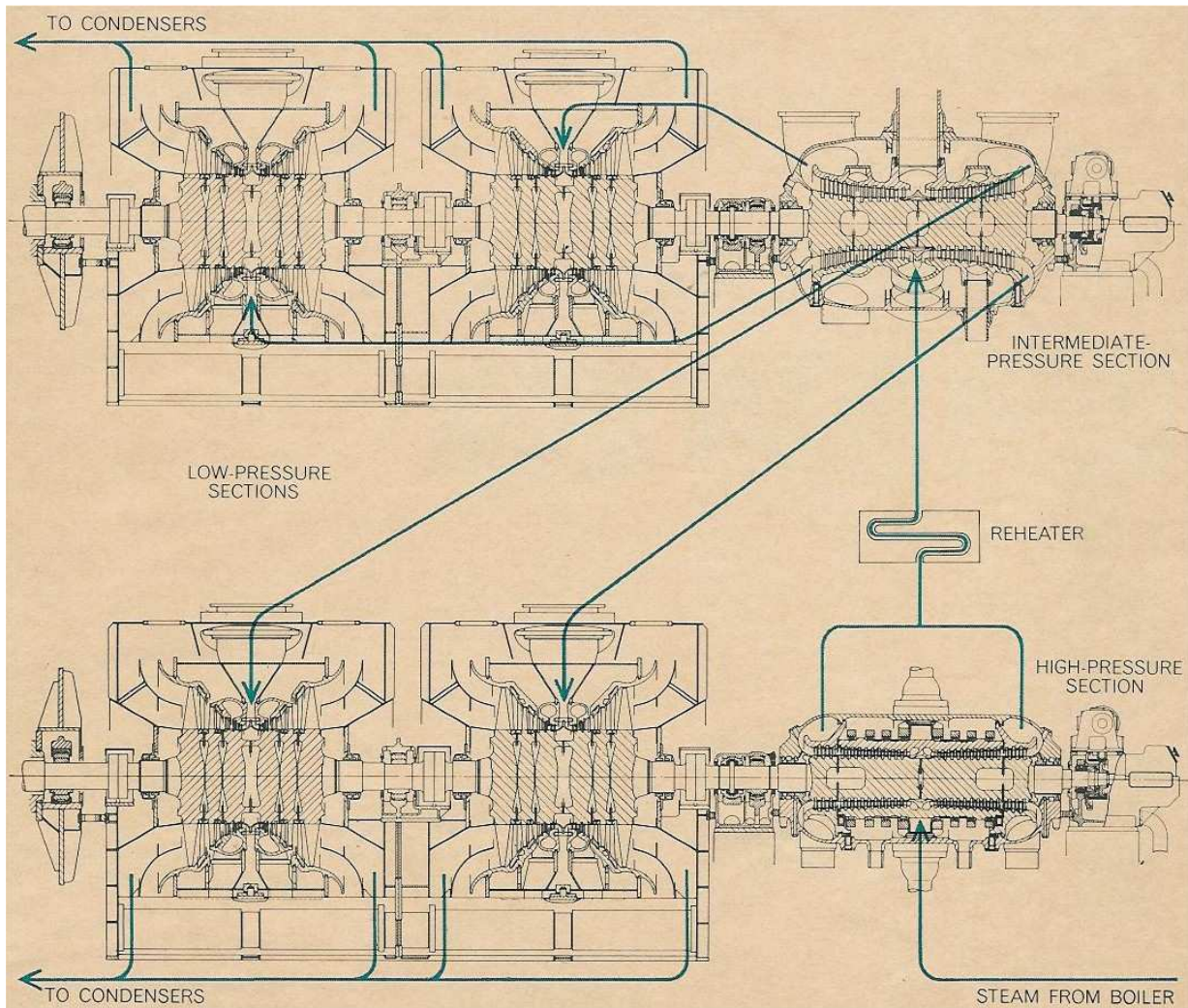
**Fig. 10**  
**PROJECTION OF TURBINE SIZES**, made by Brown, Boveri, reflects the fact that U.S. nuclear power plants (top curve) run larger than U.S. fossil-fuel plants (middle curve) and that both are larger than fossil-fuel plants contemplated in Britain (bottom curve).



**Fig. 11**

A 1,300-MEGAWATT TURBOSET, now being built by Brown, Boveri, will be the largest unit in operation when it is delivered to the TVA in 1971. It is described as a cross-compound unit, indicating that the steam travels sequentially through turbines of different sizes located on two different shafts (see Fig. 12). The turboset will run on steam produced by burning of fossil fuel.

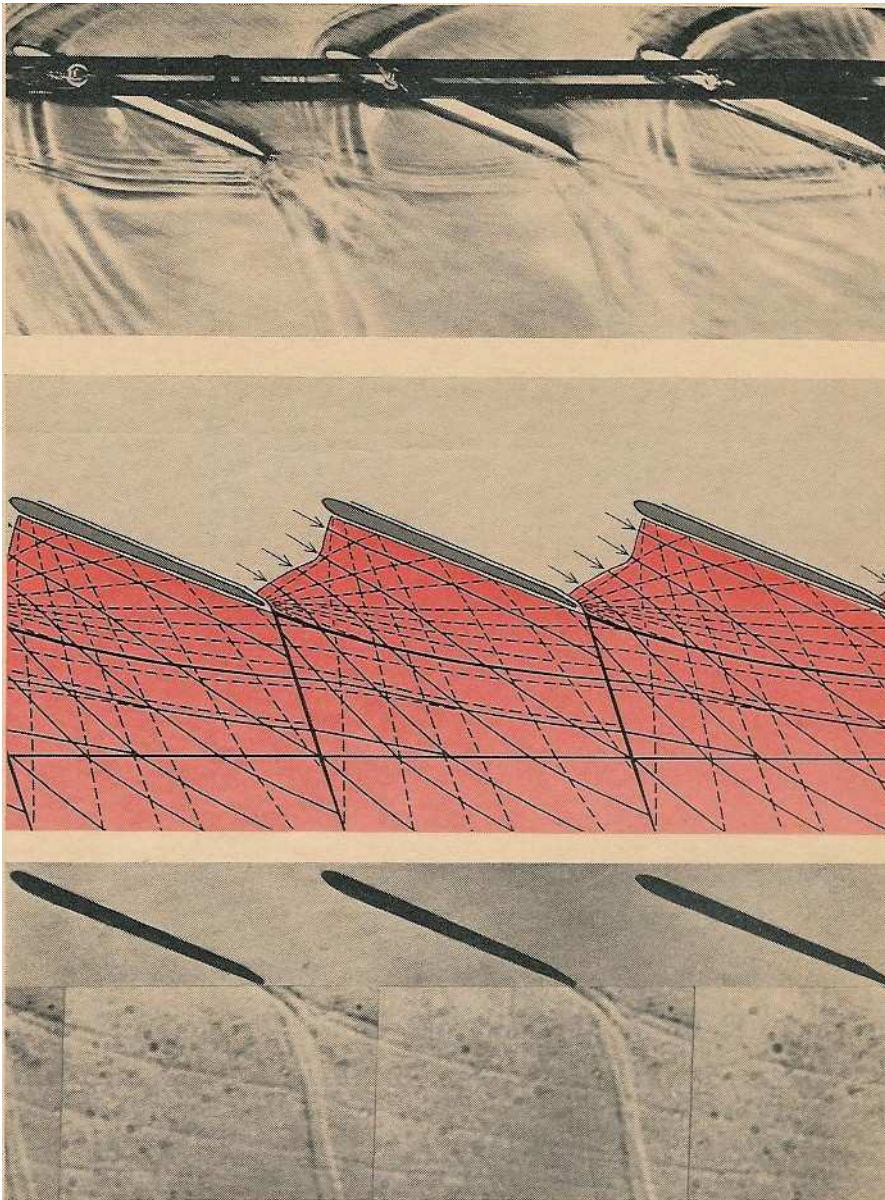




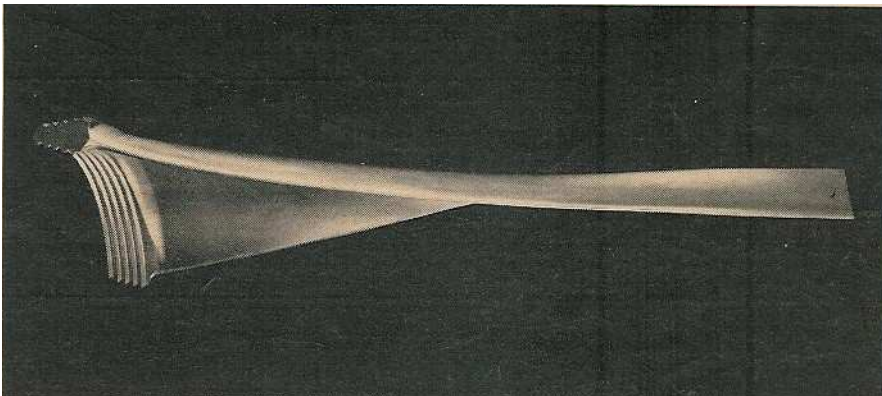
**Fig. 12**

**FLOW PLAN OF TVA TURBOSET** shows how the steam is conducted through a double-flow high-pressure turbine, a double-flow intermediate-pressure turbine and finally four double-flow low-pressure turbines, all mounted on two shafts turning at 3,600 revolutions per minute. Steam enters the turboset at 3,500 pounds per square inch and 1,000 degrees F. Before passing to the intermediate pressure turbines the steam is reheated to the same temperature. The flow of steam is 4,000 tons per hour, or 1.1 tons per second.





**Fig. 13**  
**TURBINE-BLADE**  
**CHARACTERISTICS** can be studied in three general ways: by water-tank analogy (top), by direct calculation (middle) and by airflow tests in a rotating cascade (bottom). It can be seen that the three methods give comparable results. These particular studies show patterns of transonic flow when blades are traveling at 1.4 times the speed of sound in air. In the middle picture the area tinted in color denotes supersonic flow.



**Fig. 14**  
**TYPICAL LAST-STAGE**  
**BLADE** is 95 centimeters (37.5 inches) long, excluding the root. When installed and running, the tip of the blade will travel at 600 meters per second, or 1.6 times the speed of sound in steam under the conditions existing in the turbine's last stage.





**Fig. 15**  
**RESONANCE**  
**FREQUENCIES**  
of last-stage blades can now be calculated with good accuracy. The slanting lines (1 through 8) are potential exciting frequencies. Conditions to be avoided (open dots) occur where these lines intersect the vertical line at the normal shaft speed, here 3,000 revolutions per minute. The solid lines in color ( $f_0$ ,  $f_1$ ,  $f_2$ ) are resonant frequencies of the blade as obtained by calculation. The broken lines are measured values.